

# Empirical estimation of regenerator efficiency for a low temperature differential Stirling engine



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## ABSTRACT

An experimental method of regenerator evaluation is proposed in this paper. The configuration of the experimental equipment used in the method is similar to that of an alpha-configuration Stirling engine with a phase angle of  $180^\circ$ . The temperature of the hot side heat exchanger is controlled by an electric heater, and the heat sink was room air. An air conditioner controlled the temperature of the room air. The temperature and pressure of the working fluid were measured during the piston motion. A #18 stainless steel mesh was used as a regenerator matrix for a low temperature differential Stirling engine (LTDSE). The regenerator efficiency can be calculated based on the measurement results. The product of the swept volume, the density of the working fluid, the specific heat and the difference in the working fluid temperatures between the hot side and the cold side is greater than the amount of the internal energy fluctuation. The reason for this is assumed to be the temperature fluctuation in the region between the two heat exchangers. The walls of the region are made of acrylic resin. The amount of the temperature fluctuation in the region is assumed to be uniform. The regenerator efficiency is calculated as a function of the temperature fluctuation in the region. The evaluation method does not require a fast-response thermocouple. The prediction of the regenerator efficiency is possible based on some experimental results of same matrix. Polyurethane foam and #18 stainless steel mesh, layered parallel to the stream line of the working fluid, were each tested. These materials can realize a non-rectangular regenerator without the generation of waste. Non-rectangular regenerator includes regenerator that can be installed into narrow gaps. The regenerator efficiency of the stainless steel mesh layered parallel to the stream line of the working fluid was significantly less in comparison to that of the normal mesh layers. In the polyurethane foam case, a pressure loss was observed.

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## 1. Introduction

The purpose of the study described in this paper is to develop a desktop 100 W level low temperature differential Stirling engine (LTDSE) with a heat source temperature below  $100^\circ\text{C}$ . A large number of low temperature heat sources exist. A utilization of such heat source may reduce some pollution in air and water, comparing with utilization of a heat source derived from combustion. However, different heat sources require different conditions, and the amount of heat produced may be small. Therefore, technology for low temperature utilization must be considered for each situation.

*Abbreviations:* LTDSE, low temperature differential Stirling engine; SE, Stirling engine.

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The motivation for the development of this system is provision of distributed power generation equipment for which maintenance is possible in remote areas.

Although an LTDSE can generate brake power from a heat source for which exergy is low, the poor power output prevents the practical use of LTDSEs. The first LTDSE was presented at the International University Center in Dubrovnik in 1983 [1,2]. LTDSE have been researched and developed over a shorter period of time than conventional Stirling engine (SE) operating with high temperature heat sources. Kongtragool and Wongwises [3–5] reported experimental data on LTDSEs. Schleder and Zoppke [6] reported on a practical LTDSE called “Sunwell50”. The SE worked as a water pump, and the energy source was sunlight. Hoshino and Yoshihara [7] reported on a free piston SE with an expansion space temperature of  $100^\circ\text{C}$ .

Baba and Kato [8] presented the indicated diagrams obtained by low temperature differential Stirling engine operations. Kato and Oba [9] performed numerical calculations using SCM20 [10], which

### Nomenclature

|            |                                                                         |
|------------|-------------------------------------------------------------------------|
| $\alpha_r$ | heat transfer coefficient between the regenerator and the working fluid |
| $\lambda$  | thermal conductivity of the working fluid                               |
| $\eta_r$   | regenerator efficiency                                                  |
| $\nu$      | kinetic viscosity                                                       |
| $\rho$     | density of the working fluid                                            |
| $A_{pp}$   | cross-sectional area of the piston                                      |
| $C_i$      | coefficient, where $i$ is an integer                                    |
| $C$        | specific heat of the working fluid                                      |
| $d_i$      | exponent, where $i$ is an integer                                       |
| $d_m$      | wire diameter                                                           |
| $E_{tu}$   | the number of heat transfer units                                       |
| $Nu$       | Nusselt number                                                          |
| $\Delta P$ | pressure fluctuation of the working fluid during one cycle              |

|                  |                                                                                                   |
|------------------|---------------------------------------------------------------------------------------------------|
| $R$              | gas constant                                                                                      |
| $S_r$            | surface area of the regenerator                                                                   |
| $\Delta T_{ins}$ | temperature fluctuation in the space of the measurement sections and regenerator during one cycle |
| $T_c$            | mean temperature of the cold side working fluid                                                   |
| $T_{ea}$         | ensemble average of the temperature of the working fluid in equipment                             |
| $T_h$            | mean temperature of the hot side working fluid                                                    |
| $u$              | velocity of the working fluid                                                                     |
| $\Delta U_{exp}$ | internal energy fluctuation of the working fluid during one cycle                                 |
| $V_m$            | bulk volume of one side of the measurement section                                                |
| $V_r$            | bulk volume of the regenerator                                                                    |
| $V_T$            | total volume of the working fluid in the equipment                                                |
| $x_{st}$         | length of the piston stroke                                                                       |

is a numerical simulation program for the design of Stirling cycle machines, and suggested that the utilization of a regenerator or the enhancement of the heat exchange would improve the performance of LTDSEs.

This paper discusses an LTDSE in which the inner surfaces of the displacer chamber work as a heat exchanger. As Fig. 1 shows, there are two types of heat exchanger installations in LTDSEs. When a part of the displacer chamber works as a heat exchanger, the gap between the displacer side wall and the wall of the displacer chamber can be a path for the working fluid. This type of LTDSE requires seals only at the power piston and the displacer rod. Eliminating the external heat exchanger such as shown in right side of Fig. 1 increases the compression ratio. It was thought that the path in a regenerator for such an LTDSE must be significantly larger than the gap between the displacer's inner wall and the side surface of the displacer. Therefore, a regenerator matrix with a pitch of approximately 1 mm is suitable.

As Yamashita et al. [10] showed, many studies have been carried out on regenerators. However, there is no convenient method for determining the geometry of a regenerator in a new Stirling engine design. The prediction of regenerator performance is difficult. Otaka and Ota [11] described a method that required control of the gas flow rate and temperature. Tanaka et al. [12] described method that required fast response thermocouple. Both methods are inconvenient. The improvement of LTDSEs requires trial and error. Therefore, a more convenient method for regenerator evaluation is required.

Additionally, wasted regenerator material occurs when the cross-sectional area of the regenerator is not rectangular. If the metal mesh matrixes which are layered parallel to the stream lines of the working fluid can function the same as those that are layered

normally, a non-rectangular cross-section can be made without wasting any mesh. For example, a roll of metal mesh matrix can be formed into a cylindrical regenerator. Additionally, a metal mesh matrix can be installed within a narrow gap. Polyurethane foam also is candidate regenerator material, capable of realizing non-rectangular regenerator shapes without generating waste.

This paper describes the effect of #18 stainless steel mesh as the regenerator matrix. The opening area of the mesh is larger than the gap between the displacer and the displacer chamber wall in an LTDSE. This paper also proposes a convenient experimental method for regenerator evaluation and a method for prediction of regenerator efficiency. Additionally, the regenerator which can be installed into a narrow gap is of interest. Because a material that can be used to construct a regenerator with a non-rectangular cross-section without the generation of waste was desired, both polyurethane foam and #18 stainless steel mesh, layered parallel to the stream line of the working fluid, were also tested. These materials are hard to stain. Being hard to stain is advantageous to use as a regenerator. The regenerator efficiency evaluations also were carried out in these regenerators.

## 2. Methods

### 2.1. Overview

Evaluations of the effect of #18 stainless steel mesh as the regenerator matrix were carried out. The same data were used for the discussion of a convenient experimental method for regenerator evaluation and a method for prediction of regenerator efficiency.

Evaluations of the metal mesh layered parallel to the steam line of the working fluid and the polyurethane foam were carried out using the new proposed method. The same experimental equipment was used for all the experiments in this paper. However, a part of specifications of the experimental equipment was modified when the data for comparison of three kinds of the regenerators were obtained. The regenerator consisting of the normally layered stainless steel matrixes was tested using both the original experimental equipment and the modified experimental equipment.

### 2.2. Experimental apparatus

Fig. 2 shows a schematic view of the experimental equipment. Although the configuration of the experimental equipment is

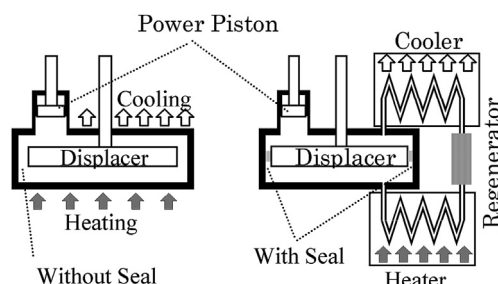


Fig. 1. Models of Stirling engines compared.

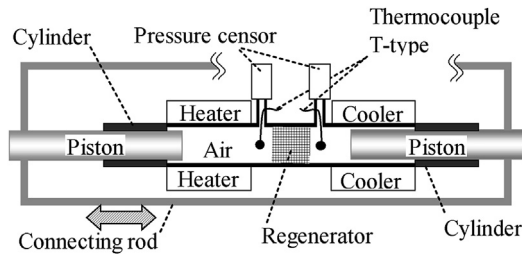


Fig. 2. Schematic view of the experimental equipment.

similar to that of an alpha-configuration Stirling engine, the equipment is not Stirling engine. Electric motor caused the motions of two pistons. To avoid leaking of the working fluid from the sliding surfaces between the pistons and cylinder liners, the pistons and cylinders were made of syringes produced by the Tsubasa Industry Co., Ltd. Except for the sliding surfaces, the components of the power pistons were made of acrylic resin. The two heat exchangers and the piston holders were made of aluminum. In the section between the hot side and the cold side, both the heat loss from inside to outside and the heat transfer through the walls from the hot side to the cold side should be minimized. Therefore, all the walls of the section were fabricated with acrylic resin parts. An acrylic resin spacer was installed between the cylinder and the heat exchanger on the hot side.

Table 1 shows the specifications of the original equipment, which was used for the evaluations of the effect of #18 stainless steel mesh as the regenerator matrix. The phase angle of the two pistons was 180°. The total volume of the working fluid in the original equipment was a constant 64.4 cc. The working fluid was air. The mean pressure of the working fluid was atmospheric pressure. The volume of the section for the regenerator was 8.8 cc. The total volume of the sections for measurements of pressure and temperature was 6.3 cc. Measurement sections were present on both the hot side and the cold side. Gaps existed between the pistons and the heat exchangers. The total volume of the gaps was 1.3 cc. Piston strokes of 20, 40 and 60 mm, with corresponding stroke volumes of 11.9, 23.8 and 35.6 cc, respectively, were used in the experiments.

Table 2 shows the specifications of the modified equipment, which was used to compare the three kinds of regenerators. The total volume of the working fluid in the modified equipment was a constant 60.7 cc. Piston strokes of 30, 40, 50 and 60 mm were used in the experiments.

In this paper, the term “heat exchanger” denotes the heat exchanger between the working fluid and the heat source or heat

Table 2

The specifications of the modified experimental equipment for regenerator comparison.

|                                                                              |                |
|------------------------------------------------------------------------------|----------------|
| Diameter of the pistons [mm]                                                 | 27.5           |
| Stroke of the pistons [mm]                                                   | 30, 40, 50, 60 |
| Inner diameter of the heat exchangers [mm]                                   | 28.0           |
| Length of the hot side exchanger [mm] <sup>a</sup>                           | 73             |
| Length of the cold side exchanger [mm] <sup>b</sup>                          | 66             |
| Phase angle [degree]                                                         | 180            |
| Distance between the two piston tops [mm]                                    | 114            |
| Space for the regenerator [mm]                                               | 20 × 20 × 22   |
| Inner diameter of the spaces for measurement of the pressure and temperature | φ20            |
| Length of the spaces for measurement of the pressure and temperature         | 11             |

<sup>a</sup> The length includes the thickness of the gaskets and the plastic piece between the heat exchanger and the cylinder.

<sup>b</sup> The length includes the thickness of the gaskets.

sink. Although the regenerator is a heat exchanger, the word “heat exchanger” does not apply to the regenerator in this paper.

### 2.3. Measurements

In the experiments, pressures and temperatures were measured during piston movement. During the experiments, the temperature of the hot side heat exchanger was controlled using an electric heater. The heat sink was atmospheric air. The temperature of the air was controlled using an air conditioner.

The inner pressures of both the hot side and the cold side were measured. The pressure sensor heads were Keyence AP-41A, which have a rated pressure range from −101.3 to 101.3 kPa. The resolution in standard mode was 0.1 kPa, and the response was 1 ms. The amplifier for the pressure sensor head was a Keyence AP-44. A USB oscilloscope Picoscope4424 was used as an A/D converter to send the data from the pressure and crank angle to a personal computer. The resolution of the pressure measurement was 0.167 kPa because of the resolution of the A/D converter. The analog output from the AP-44 was from 1 to 4 V, and the range of the analog output was correlated with the rated pressure range of the AP-41A. The resolution of the 12 bit oscilloscope in the range of −5 to 5 V is approximately 0.0033 V, which becomes approximately 0.167 kPa. The pressure data were processed by a software low-pass filter of the PicoScope when the pressure was measured. The experiments to determine the filtered frequency were conducted in advance. To obtain the data for the crank angle, a Keyence FU, which consists of a digital fiber sensor FS-N10 Series and an FS-N11N, was used. The A/D converter for the pressure measurement was also used for the crank angle.

For the temperature measurement, a USB TC-08 thermocouple data logger provided by Pico Technology sent the data from the thermocouples to a personal computer. In the original equipment, type T thermocouples with a diameter of 0.1 mm were used to measure the temperature of the working fluid. Type K thermocouples with a diameter of 0.3 mm measured the outer surface temperatures of the heat exchangers and the atmospheric temperature. The USB TC-08 thermocouple data logger and the software recorded five temperatures of the thermocouples and the temperature of the thermistor in the TC-08 thermocouple data logger every 0.6 s. The error of the thermocouple was less than 0.6 °C when all of the thermocouples and the thermistor were compared. In the modified equipment for regenerator comparison, type T thermocouples with a diameter of 0.3 mm were used to measure the temperature of the working fluid. A thermistor was engaged in the USB TC-08 thermocouple data logger to measure the atmospheric temperature.

Table 1

The specifications of the original experimental equipment.

|                                                                              |              |
|------------------------------------------------------------------------------|--------------|
| Diameter of pistons [mm]                                                     | 27.5         |
| Stroke of pistons [mm]                                                       | 20, 40, 60   |
| Inner diameter of heat exchangers [mm]                                       | 28.0         |
| Length of hot side exchanger [mm] <sup>a</sup>                               | 73           |
| Length of cold side exchanger [mm] <sup>b</sup>                              | 66           |
| Phase angle [degree]                                                         | 180          |
| Distance of two piston tops [mm]                                             | 120          |
| Space for regenerator [mm]                                                   | 20 × 20 × 22 |
| Inner diameter of the spaces for measurement of the pressure and temperature | φ20          |
| Length of the spaces for measurement of the pressure and temperature         | 11           |

<sup>a</sup> The length includes the thickness of the gaskets and the plastic piece between the heat exchanger and the cylinder.

<sup>b</sup> The length includes the thickness of the gaskets.



**Table 3**

The data for the #18 stainless steel mesh regenerator matrix.

|                                                     |             |
|-----------------------------------------------------|-------------|
| Diameter of the wire $d_m$ [mm]                     | 0.2         |
| Pitch $p$ [mm]                                      | 1.41        |
| Width of the opening area $l$ [mm]                  | 1.21        |
| Porosity $\phi$                                     | 0.888       |
| Specific surface area [ $\text{mm}^2/\text{mm}^3$ ] | 2.09        |
| Bulk volume of 1 matrix [ $\text{mm}^2$ ]           | 80          |
| Height, width, thickness of 1 matrix [mm]           | 20, 20, 0.2 |
| Opening area ratio $\beta$                          | 0.737       |

#### 2.4. Regenerator

Table 3 shows the specifications of the regenerator matrix. The numbers of regenerator matrixes installed in the original equipment for #18 stainless steel matrix evaluation were 0, 15, 30 and 60. The heat capacity of one matrix was 0.281 J/K. The heat capacities of 20 matrixes, 40 matrixes and 60 matrixes are, respectively, 5.6, 11.2 and 16.8 J/K. The properties of the matrix were assumed to be the same as that of an austenitic stainless steel at 300 K. The density of such a material is 7920 kg/m<sup>3</sup>, and the specific heat is 0.499 kJ/(kg K) [13].

Fig. 3 shows the polyurethane foam, #18 stainless steel mesh. The sizes of the pores in polyurethane foam are not uniform. Fig. 4 shows the cross-section of the regenerator consisting of the matrixes layered parallel to the stream line of the working fluid. The matrixes were the same as that in the regenerator consisting of the normal layered matrixes. Fig. 4 shows that the wires are significantly denser than in the normal layered #18 matrixes and that the porosity is not uniform. The regenerator consisting of the normal layered matrixes was composed of 60 sheets of matrixes to compare the three kinds of regenerators. The regenerator consisting of the matrixes layered parallel to the stream line of the working fluid was composed of 57 sheets of matrixes in the present experiments. For the case in which the matrixes were layered parallel to the stream line of the working fluid, the number of installed matrixes was different in each installation, varied by a few sheets in each case. Although the experimental result was different for each installation, the difference is completely negligible in the following discussion.

### 3. Experimental results and discussion

#### 3.1. Characteristic of #18 stainless steel mesh regenerator matrix

During the experiments, the room temperature was approximately 20 °C. The hot side heat exchanger temperature varied from



Fig. 4. The stainless steel meshes layered parallel to the stream line direction of the working fluid.

60 to 63 °C. The cold side heat exchanger temperature varied from 21 to 23 °C. No significant pressure difference between the hot side and the cold side was observed during the experiments. The temperature difference between the heat exchanger and the working fluid on the hot side varied from 10 to 15 °C. The temperature difference on the cold side varied from 3 to 11 °C. On the hot side, heat radiating from the heat exchanger to atmospheric air affected the temperature difference between the heat exchanger and the working fluid.

The pressure fluctuation of the working fluid during one cycle  $\Delta P$  was determined for each experiment. Eq. (1) and Eq. (2) yield Eq. (3), with which the internal energy fluctuation is calculated. In Fig. 5, the internal energy fluctuations divided by the specific heat of the working fluid, the stroke volume of the piston and the temperature difference between two heat exchangers are expressed as dimensionless values. Fig. 5 suggests that the #18 stainless steel mesh matrix improves an LTDSE. The dimensionless values obtained from the experiments with the regenerators are more than double the values obtained without the mesh. The

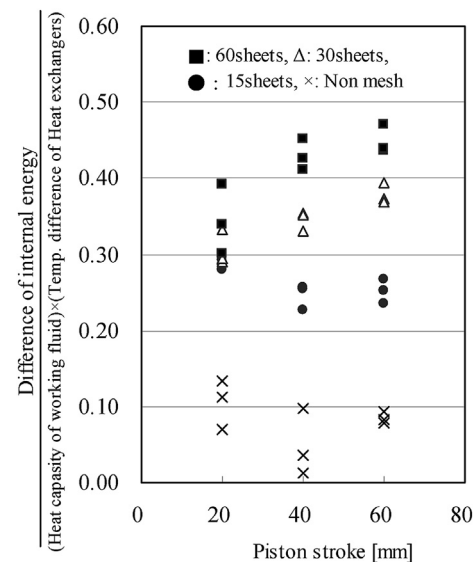


Fig. 5. Fluctuation of the internal energy during 1 cycle.



Fig. 3. The polyurethane foam, #18 stainless steel mesh and scale with a distance of 1 mm.

dimensionless values for the non-mesh conditions were approximately 0.1. This value, 0.1, is almost same as the empirical value for the design of a model LTDSE. For non-mesh conditions, the experimental results were reasonable.

$$\Delta P = \rho R \Delta T_{ea} \quad (1)$$

$$\Delta U_{exp} = \rho V_T C \Delta T_{ea} \quad (2)$$

$$\Delta U_{exp} = V_T \frac{C}{R} \Delta P \quad (3)$$

As Fig. 5 shows, when the piston stroke was 20 mm, the number of matrixes had no significant effect. In addition, the effect of the engine speed was not significant compared with other experimental parameters such as the number of the matrixes and the piston stroke. The number of matrixes was positively correlated with the dimensionless internal energy fluctuation. However, the effect was not proportional. Increasing the piston stroke increased the dimensionless internal energy fluctuation. In addition, Fig. 5 suggests that the factors this paper does not discussed will affect on the performance of LTDSE.

The effect of stroke in the experiments with 15 matrixes was different from that with other numbers of matrixes. When the number of matrixes was 15 and the piston stroke was 40 or 60 mm, the ratio of regenerator heat capacity to working fluid heat capacity was lower than the difference in the temperature of the working fluid on the hot side and that on the cold side.

As Fig. 6 shows, a low stroke volume or a large number of regenerator matrixes decreases the temperature fluctuation. The trends illustrated by Fig. 6 suggest that #18 stainless steel mesh works as a regenerator matrix. The presence of #18 stainless steel mesh affected the temperature difference between the hot-side working fluid and the cold-side working fluid. When the stainless steel mesh was not present, the temperature difference varied from 9 to 19 °C. When the stainless steel mesh was present, the temperature difference was approximately 23 °C, except for some experiments.

### 3.2. Evaluation of regenerator efficiency for LTDSE design

The internal energy fluctuation described above consists of heat exchange by two heat exchangers and the effect of a regenerator. As shown in Fig. 5, the effect of two heat exchangers on the internal

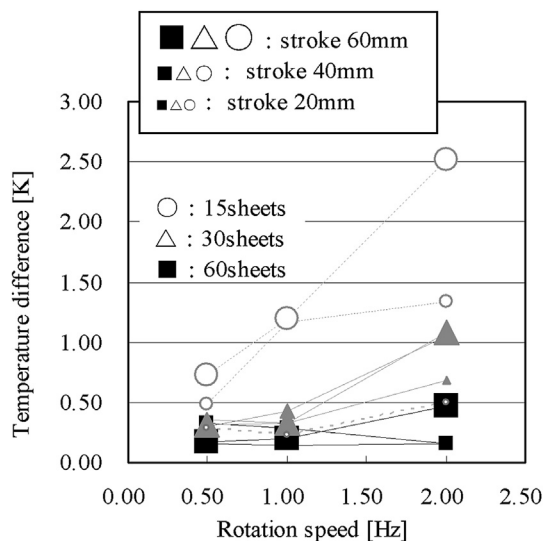


Fig. 6. Fluctuation of the temperature of the cold side working fluid.

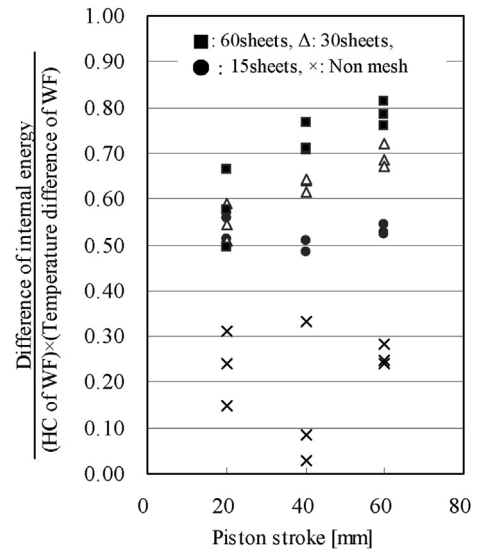


Fig. 7. Internal energy nondimensionalized by the temperature of the working fluid.

energy fluctuation is not negligible. Therefore, the regenerator efficiency  $\eta_r$  was calculated as described below.

As Fig. 7 shows, the ratios of the internal energy fluctuations to the heat capacity of the working fluid and the temperature difference of working fluid were less than 1. This is thought to be the reason that the temperature distribution in the section surrounded by the acrylic resin walls changes as shown in Fig. 8. In Fig. 8, the temperature of the measurement section of the inlet side is same as the temperature of the adjacent heat exchanger. On the outlet side, the temperature of the measurement section is the temperature of the regenerator outlet.

$$\rho C \cdot A_{pp} x_{st} (T_h - T_c) = \Delta U_{exp} + \rho C (V_r + 2V_m) \Delta T_{ins} \quad (4)$$

$$\eta_r = 1 - \frac{\Delta T_{ins}}{T_h - T_c} \quad (5)$$

$$\eta_r = 1 - \frac{2}{E_{tu} + 2} \quad (6)$$

$$E_{tu} = \frac{\alpha_r S_r}{\rho C \cdot A_{pp} x_{st}} \quad (7)$$

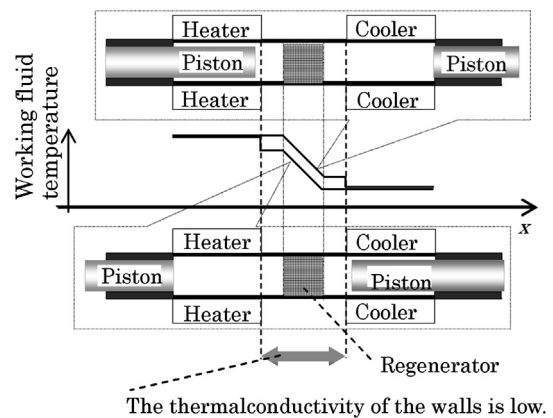


Fig. 8. Behavior of the temperature in the space surrounded by the low-thermal-conductivity walls.

$$Nu = \frac{d_m \alpha_r}{\lambda} \quad (8)$$

$$Nu = C_1 Re^{d_1} \quad (9)$$

$$Re = \frac{d_m u}{\nu} \quad (10)$$

$$E_{tu} = C_1 \frac{\lambda}{\rho \nu^{d_1} C A_{pp} x_{st}} \frac{S_r}{d_m^{1-d_1}} u^{d_1} \quad (11)$$

$$E_{tu} = C_2 \frac{S_r}{A_{pp} x_{st}} u^{d_1} \quad (12)$$

The ensemble average of temperature fluctuation in the section surrounded by the acrylic resin walls is assumed to be uniformly  $\Delta T_{ins}$ . Based on this assumption,  $\rho(V_r + 2V_m)C\Delta T_{ins}$  is the fluctuation of the internal energy in the space of the regenerator and both sides of the measurement sections. Here, the energy balance is assumed to be expressed by Eq. (4). In addition, the regenerator efficiency  $\eta_r$  is assumed to be expressed by Eq. (5).  $\Delta T_{ins}$  is the temperature fluctuation in measurement sections and regenerator section, and is obtained in each experimental condition. Eq. (6) is the equation that Yamashita et al. [10] quoted from the work of Hausen [14], which expresses the relation between the regenerator efficiency  $\eta_r$  and the number of heat transfer units  $E_{tu}$ , when the heat capacity of the regenerator is sufficiently large compared to the heat capacity of the working fluid. In this paper, the number of heat transfer units  $E_{tu}$  is defined by Eq. (7). The heat transfer coefficient  $\alpha_r$  is given by Eq. (8).

Eq. (9) expresses the form of some experimental formulae quoted by Yamashita et al. [10]. Eq. (10) expresses the Reynolds number of Eq. (9). Eqs. (7)–(10) yield Eq. (11). When the working fluid and the regenerator matrix are the same, the coefficient  $C_2$  is available, and Eq. (12) is obtained. When the regenerator matrix is the same, the surface area of the regenerator  $S_r$  is proportional to the volume of the regenerator. The working fluid velocity  $u$  is determined from the engine speed, the stroke volume and the cross-sectional area of the regenerator section. The values of the coefficient  $C_2$  and the exponent  $d_1$  are obtained based on the experimental results. Obtaining the values of the coefficient  $C_2$  and the exponent  $d_1$  makes it possible to predict the regenerator efficiency using Eq. (12) and Eq. (6).

In this study, the values of the coefficient  $C_2$  and the exponent  $d_1$  were obtained from the results of the experiments in which the number of installed regenerator matrixes was 30 or 60 and the piston stroke was 40 or 60. Fig. 9 shows a comparison of the experimental data and the predicted values in terms of the regenerator loss. In addition, Fig. 9 suggests that the temperature difference between the heat exchanger and the working fluid on the cold side are positively correlated with the regenerator loss and the cold side working fluid temperature fluctuation. These positive correlations are reasonable. It should also be noted that the use of atmospheric air as the heat sink made it possible to neglect the heat loss from the cold side heat exchanger to the surroundings.

As shown in Fig. 8, the estimation method in this paper is based on the assumption that the temperatures measured by the thermocouples were the temperatures in the heat exchangers. When the temperature measured by the thermocouple is assumed to be a mean temperature, the temperature of the working fluid in the hot side heat exchanger is a summation of the indicated thermocouple value and the half of the temperature fluctuation in the regenerator  $\Delta T_{ins}$ . Additionally, the temperature of the working fluid in the cold side heat exchanger is the difference of the indicated thermocouple

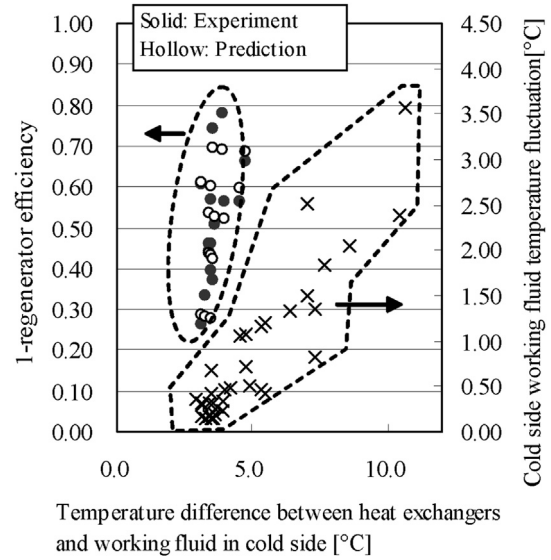


Fig. 9. Regenerator loss obtained using Eq. (5), Regenerator loss obtained using Eqs. (6) and (12) and the cold side working fluid temperature fluctuation.

value and half of the temperature fluctuation in the regenerator  $\Delta T_{ins}$ . This assumption causes a significantly increased regenerator efficiency in comparison to that of the assumption shown in Fig. 8. However, this assumption often causes a greater difference in the working fluid temperatures between the two heat exchangers than the difference in the outer surface temperatures between the two heat exchangers. Although the assumption shown in Fig. 8 estimates the regenerator efficiency to be less than the actual regenerator efficiency when the regenerator efficiency of the sample material is low, an accurate estimation is not necessary in the regenerator material of which regenerator efficiency is low. Further discussion of such a material will not improve the LTDSE performance.

### 3.3. The effects of piston stroke

As shown in Fig. 10, the pressure oscillation, which is proportional to the internal energy fluctuation, correlates with the piston stroke. However, the correlation between the piston stroke and the regenerator efficiency cannot be confirmed based on Fig. 10. As explained in the following discussion, it is thought that the piston stroke does not correlate with the regenerator efficiency in Fig. 10 because of the effect of the working fluid velocity, as caused by piston stroke, which is not considered in the method used to evaluate regenerator efficiency. Although the experimental results shown in Fig. 10 appeared to contain error, the reproducibility was confirmed. For example, in the conditions in which the regenerator consisting of the normal layered matrixes was tested, and for which the operational frequency was over 60 rpm, the pressure oscillation caused by the 60 mm piston stroke was similar to that caused by the 50 mm piston stroke. Additionally, the data obtained in the conditions with the piston stroke of 50 mm do not allow for a linear approximation. Considering the above, there must be the factor that we have not yet considered in our analysis but that also strongly affected the experimental results.

Increasing the piston stroke causes the following to occur.

It increases the amount of working fluid passing through the regenerator and reduces the regenerator efficiency. It increases the working fluid velocity. The higher velocity of the working fluid decreases the Nusselt number. It increases the amount of working



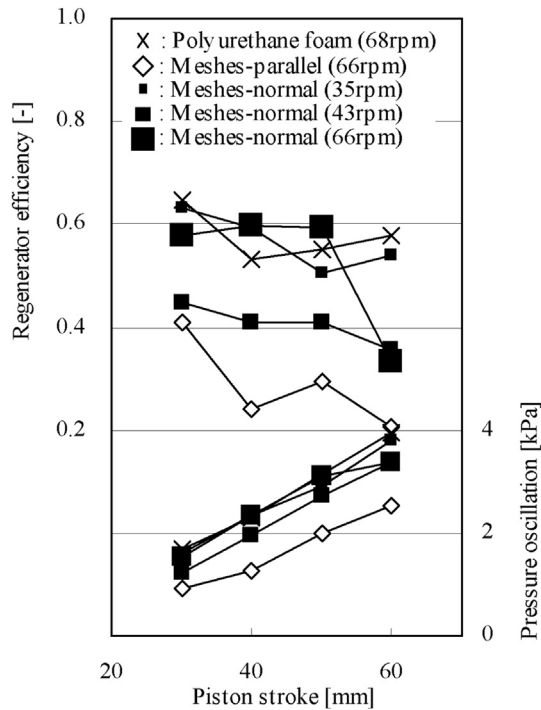


Fig. 10. The regenerator efficiencies and pressure oscillation in each material and in each piston stroke.

fluid flowing into the heat exchanger section. The increase in the amount of working fluid flowing into the heat exchanger section comparatively decreases the effect of the space between the regenerator and heat exchanger, which is bulk without significant heat exchange. Most of the above mentioned issues decrease the regenerator efficiency. However, the effect of each issue on the heat transfer in the heat exchanger is different for each case, in addition to being complicated. Considerations of the issues are insufficient for discussing the effects of the piston stroke and operating frequency.

#### 3.4. Evaluation of the regenerator consisting of the matrixes layered parallel to the stream line

As shown in Fig. 10, the regenerator efficiencies of the regenerator consisting of the matrixes layered parallel to the stream line of the working fluid are less than that of the regenerator consisting of the normal layered matrixes. The differences in the surface areas and heat capacities are a few percent. It is clear that the difference in the regenerator efficiency was not caused by a difference of only a few sheets of matrixes. The specific installation affected the regenerator efficiency. Although the heterogeneous porosity shown in Fig. 4 also may decrease the regenerator efficiency, it was not confirmed experimentally in this study.

#### 3.5. The evaluation of the polyurethane foam

Fig. 10 shows the representative data chosen for comparison of the regenerator materials. In the experiments using polyurethane foam, the pressure oscillations were greater, and the temperature difference between the hot side working fluid and the cold side working fluid also were greater, in comparison to the data obtained from the experiments using the regenerator consisting of the normal layered stainless steel matrixes. However, the regenerator efficiency of the polyurethane foam was less than of the

regenerator consisting of the normally layered stainless steel matrixes. The combination of greater pressure oscillation, higher temperature difference and reduced regeneration efficiency is caused by the pressure loss of the polyurethane foam, if such a pressure loss in the polyurethane foam existed. Pressure loss prevents the working fluid flow. Plugging the flow maintains the temperature difference between the two sections. Additionally, plugging the flow causes a volume expansion without material flow. Considering the equation of state for an ideal gas in the hot side heat exchanger, the gas constant is clearly constant, and the volume is determined by the position of the piston. The combination of higher temperature and higher pressure gives the reduced mass of the working fluid. Although the combination of the greater temperature difference and greater pressure oscillation may associate with high performance, the greater pressure loss causes the combination of the greater temperature difference and greater pressure oscillation. The greater pressure loss must be distinguished from the higher regenerator efficiency. In this study, the pressure oscillation in the operation using polyurethane foam was the same as that using the regenerator consisting of the normal layered stainless steel matrixes. Therefore, it was easy to conclude that the regenerator consisting of the regenerator consisting of the normal layered stainless steel matrixes was better.

As shown in Fig. 11, the time at which the extreme values of pressure were obtained varied between the hot side and the cold side, when the polyurethane foam was tested under the operating frequency of 68 rpm with a piston stroke of 60 mm. The difference in time at which the extreme values of pressure were obtained between the hot side and the cold side also suggests the pressure loss in the polyurethane foam must not be negligible.

From Fig. 11, the maximum obtained difference in pressure between the hot side and the cold side is 0.25 kPa. The value is less than the noise of signal. Although the accuracy is insufficient, the value suggests that the tested polyurethane foam cannot function as a regenerator under operating conditions in which the working fluid velocity is similar to that of the conditions shown in Fig. 11, when the gap between the displacer and the displacer chamber is greater than 0.1 mm. The suggestion was made following the calculation results of the distance for the two parallel plates, which caused a pressure loss of 0.25 kPa. In this calculation, the length of the two plates placed parallel to the direction of the working fluid flow is 20 mm, and the working fluid velocity is assumed to be similar to that specified in the condition shown in Fig. 11. The plate is assumed to be immobile, and the variation in the working fluid velocity is ignored. Therefore, the above mentioned value, 0.1 mm,

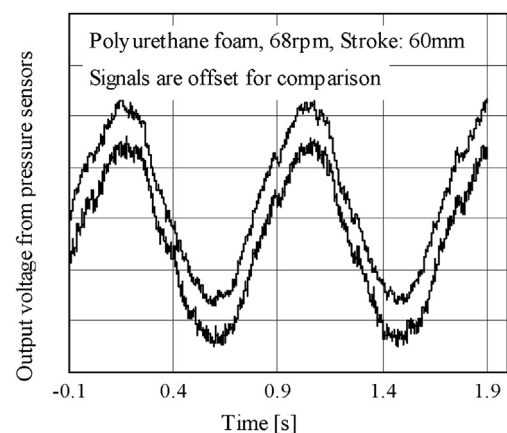


Fig. 11. The pressure difference between the hot side and the cold side in the experiment in which the sample was polyurethane foam.

is the roughly estimated value. As shown in Fig. 3, the size of the pores in the investigated polyurethane foam appears to be greater than 0.1 mm. This suggests that the gap becomes a primary stream or bypass when the width of the gap is greater or similar to that of the size of pores in the regenerator.

#### 4. Conclusions

An experimental method for regenerator evaluation was developed in this study. The configuration of the experimental equipment used in this study was similar to that of an alpha configuration Stirling engine with a phase angle of 180°. The heat sink was the surrounding air, the temperature of which was controlled using an air conditioner. A #18 stainless steel mesh functions as regenerator for an LTDSE. The following conclusions were drawn from the experimental results.

The efficiency of the regenerator depends on the difference between the temperature of the working fluid on the hot side and that on the cold side and the fluctuation of pressure. A fast response is not required in the temperature measurements.

The regenerator efficiency can be predicted from the results of experiments obtained using the same regenerator matrix. This method of predicting the regenerator efficiency is applicable to other regenerator materials. The predicted regenerator efficiency can be used in design.

However, the correlation between the piston stroke and the regenerator efficiency cannot be confirmed in the proposed method of regenerator evaluation. There must be the factor that we have not yet considered in our analysis but that also strongly affected the experimental results.

This paper also presents a discussion regarding regenerator materials capable of realizing a non-rectangular regenerator without the generation of waste. The regenerator efficiency of the stainless steel meshes layered parallel to the stream line of the working fluid was significantly less than that of normal layered meshes. For the polyurethane foam case, the pressure loss could not be ignored. Utilization of polyurethane foam as a regenerator did not allow the gap between the displacer and the displacer chamber wall to be greater than 0.1 mm. Other materials should be tested for the development of a regenerator that can be installed into narrow gap.

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